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Abstract
A team of engineers at NASA/MSFC and Boeing, Rocketdyne division, are developing unshrouded impeller technologies that will increase payload and decrease cost of future reusable launch vehicles. Using the latest analytical techniques and experimental data, a two-stage unshrouded fuel pump is being designed that will meet the performance requirements of a three-stage shrouded pump. Benefits of the new pump include lower manufacturing costs, reduced weight, and increased payload to orbit.

Introduction
In turbopump-fed rocket engines, liquid propellants are fed under high pressure into a thrust chamber where they are ignited. Thrust is developed when the hot combustion gases are expanded in the nozzle and accelerated at high velocities. Because of the high pressure in the combustion chamber, a turbopump is necessary to deliver the propellants at the required pressure. A turbopump develops the required high pressure, or "head", by spinning very fast. The faster the pump rotates, the more head is generated.

Typically, pump impellers in rocket turbopumps are shrouded. A shroud is a heavy metal casing which covers the impeller blade passages to help shape the flow, maintain performance, and develop the desired head. However, a shroud adds weight and manufacturing complexity. As a pump spins faster, stress due to centrifugal force in the impeller increases. The weight of the shroud increases the stress and limits the speed at which a pump can operate. A pump impeller without a shroud can operate at higher speeds with lower stress and generate more head.

NASA's 2nd Generation RLV Program goals are to develop safe, affordable and reliable Reusable Launch Vehicles (RLV's). Specifically, NASA will improve the safety of 2nd generation systems by two orders of magnitude (equivalent to a crew risk of 1 in 10,000 missions) and decrease the cost tenfold to approximately $1000 per pound of payload. To decrease cost, the RLV will require higher thrust-to-weight (T/W) ratio engines than currently available. One key technology that will enable significant improvements in T/W ratio and help NASA reach its goals is the application and use of unshrouded impellers.

The Marshall Space Flight Center (MSFC) has been developing unshrouded technology to improve liquid rocket engine turbopump performance. The turbopump is typically between 25% and 30% of the gross engine weight. The housing assembly makes up about 80% of the total turbopump weight. Housing size is driven by the size of the rotor assembly. Use of unshrouded impellers allows for higher tip speeds, which increases stage loading resulting in reduction of rotor and housing size and weight. This project has shown that a Space Shuttle Main Engine (SSME) 2-stage High Pressure Fuel Turbopump (HPFTP) Alternate Turbopump (AT) and a RLV YRS-2200 2-stage HPFTP would reduce turbopump weight between 45% and 50% as compared to the 3-stage designs. Table 1 illustrates the potential benefits of the increased stage loading possible with unshrouded impellers.

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Table 1 - Turbopump weight savings potential.

	Shrouded (3-stage)	Unshrouded (2-stage)	Weight Savings
SSME HPFTP/AT	990 lbs.	490 lbs.	50%
RLV HPFTP	1870 lbs.	1010 lbs.	45%

Figure 1 - Pump test equipment facility schematic.

Objective

The objective was to develop an unshrouded impeller design, using the latest analytical techniques and experimental data, which will meet the performance requirements of a 3-stage fuel pump with a 2-stage pump design[1]. This was accomplished by extending work already underway at MSFC on unshrouded impellers and applying the technology to the RLV fuel turbopump. Three major elements were used to reach the objective. The first element was to experimentally determine unshrouded impellers sensitivity to performance with tip-clearance variation and compare to analytical predictions. This was done with an SSME HPFTP/AT design point unshrouded impeller. This impeller was referred to as the Baseline Impeller. The second element was to design and test an unshrouded impeller that will meet the performance requirements of the RLV engine balance with a 2-stage pump. The design was based on experimental data and analytical techniques developed in the project. The RLV impeller is referred to as the Advanced Impeller. The third element was to produce a conceptual, complete, viable 2-stage design of the RLV fuel turbopump that incorporates the verified unshrouded impeller design.

Baseline Impeller Test

The performance of the SSME HPFTP/AT design point unshrouded impeller was experimentally verified at three tip-clearances at scaled operating conditions with water as the test fluid[2]. The purpose of the test was to extend the design database to higher stage loading supporting a reduction in RLV turbopump stage requirements.

Facility Description

The Baseline Impeller was water flow tested in MSFC's Pump Test Equipment (PTE) facility. The PTE is a closed-loop water flow facility with 10,000-gallon reservoir, deaeration and pressurization systems, facility flow meter, flow control valve, and 350 horsepower drive motor. The test facility flow schematic is shown in Figure 1.
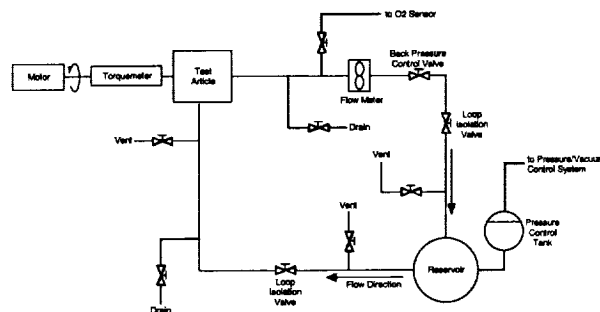
In operation flow leaves the reservoir through a 12-inch line, transitions to an 8-inch line inside the building, and enters the test article's axial inlet. An instrumentation spool just upstream of the test article inlet provides access for total pressure surveys and measurements of the pump inlet total pressure and water temperature. High pressure pump discharge flow exits through an instrumentation spool and passes through an eight inch turbine-type flow meter and back pressure valve on its way back to the reservoir. All piping is stainless steel schedule 40 with the pump discharge pressure limited to 350 psia due to the lower flow meter pressure rating. Isolation valves, drains, and vents are included where required.

Adjusting the ullage pressure in a partially filled 500-gallon tank mounted on top of and connected to the 10,000-gallon reservoir controls test article inlet pressure. The small air and water vapor volume in this pressure control tank may be pressurized with high purity air, vented to atmosphere, or evacuated with the dual vacuum pumps located in the test facility. These operations coupled with the line losses between the reservoir and test article inlet can be used to set any of a wide range of inlet pressures – from 125 psia down to 4 psia. Dissolved air was removed from the test fluid by applying a vacuum to the water-filled test loop and slowly circulating the fluid until the desired dissolved oxygen content is reached.

The PTE driveline includes a 350 horsepower 3-phase AC motor with operation from 360 to 3600 rpm via a variable speed motor controller. A Torquetronics torquemeter is located between the test article and drive motor for true torque measurement. Shear couplings between the test article and torque meter protect the device from pump shaft over-torque. A steam heat exchanger in the reservoir may be used to increase water temperatures up to 150 degrees Fahrenheit although cooling the water is not possible except through complete or partial replacement of water in the test loop with cooler fluid. Water temperature rise

during pump operation was typically small – approximately 2 degrees Fahrenheit per hour at 2700 rpm.

Shaft speed, facility flow rate, inlet total pressure, water temperature, test article pressure rise, and test article bearing temperatures are monitored at the PTE control panel by the facility operator. The operator has direct control over each of these parameters except bearing temperature and test article pressure rise and virtually any combination may be set. Table 2 summarizes the PTE operating parameters and their ranges. Limits to each are noted.

Table 2 - PTE operating envelope.

Parameter	Range
Shaft Speed	360-3600 rpm
Facility Flow Rate	300-3000 gpm
Inlet Total Pressure	4-75 psia
Pump Pressure Rise	0-250 psid
Shaft Torque	0-500 ft-lbf
Drive Line Power	0-350 hp
Water Temperature	60-160 deg F

Test Article Description

The modular design of the test article allows for use with a variety of inlet guide vanes, impellers, and diffuser configurations. A cross section of the test article appears in Figure 2.
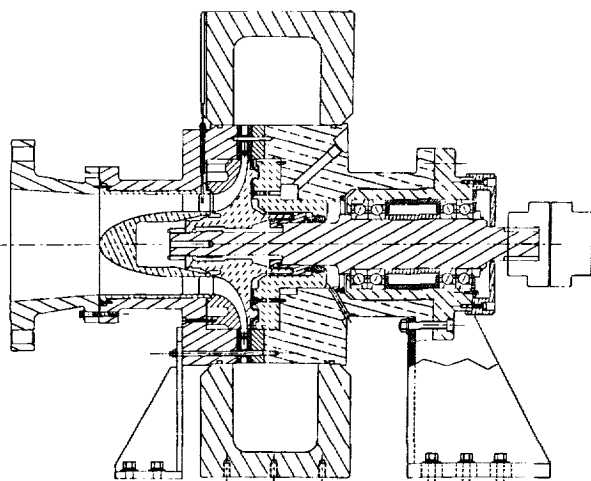

Figure 2 - Baseline unshrouded test article.

This test will combine an unshrouded version of the SSME HPFTP/AT impeller with an axial inlet and vaned radial diffuser for evaluation of impeller performance sensitivity to tip-clearance variation. The Baseline impeller has a 12-inch diameter and is 100% scale of the HPFTP/AT. The inlet guide vane assembly

includes 15 vanes with an exit blade angle of 55 degrees. The 6+6+12-impeller design, shown in Figure 3, includes 6 full-length blades, 6 medium length splitter blades, and 12 short splitter blades, resulting in 24 flow passages at the impeller discharge. The vaned radial diffuser has 23 flow passages that exit into

Figure 3 - Baseline SSME HPFTP/AT
unshrouded impeller.

generic collector volute. Rear leakage flow is collected and exits the test article through two lines. This leakage flow was metered and returned to the facility piping just upstream of the test article inlet flange. A carbon seal isolates the match bearing set from the test fluid. Five front shims were available for controlling tip-clearance for the Baseline Impeller test article. Three builds (clearance change required minor rig rebuild) were evaluated in this test series. Percent clearance is the ratio of axial clearance between the front shroud and impeller blade tip to the blade passage height (b_2). The three builds or clearances are summarized in Table 3. Build 3 was assembled without a shim for maximum clearance.

Table 3 - Baseline Impeller clearance summary.

Rig Build	Tip-Clearance	Shim ID	Percent b_2
1	0.024 inches	1	5.33%
2	0.065 inches	5	14.4%
3	0.088 inches	N/A	19.6%

Impeller blade passage height – b_2 = 0.45 inches.

Table 4 - Test Variables

Variable	Minimum	Maximum
Normalized Inlet Tip Flow Coefficient	70%	130%
Inlet Flange Suction Specific Speed*	2180 rpmxgpm$^{0.5}$/ft$^{0.75}$	30700 rpmxgpm$^{0.5}$/ft$^{.075}$
Flange Flow Rate	1047 gpm	1945 gpm
Inlet Total Pressure*	4 psia	75 psia
Leakage Flow Rate†	3 gpm	30 gpm

*Suction specific speed and inlet total pressure range based on facility capability at 2700 rpm.
†Leakage flow rate range based on measurement device capability.

Table 5 - Test matrix summary.

Description	Percent Design Flow Coefficient	Suction Specific Speed
Head-flow curve at constant suction specific speed	50% to 150%	3500
Suction performance at design flow coefficient	100%	2600 to 12120
Suction performance at higher flow coefficients	110% to 130%	2700 to 11240
Suction performance at lower flow coefficients	70% to 90%	2200 to 13680
High-frequency recording of suction and speed ramps.	70% to 130%	2200 to 13680

Test Matrix

Test article performance was evaluated over a range of scaled operating conditions at a constant shaft speed of 2700 rpm. Operationally, matching the desired range for flow coefficient and suction specific speed in Table 4 requires varying the facility flow rate and inlet total pressure at constant speed across the values shown. Impeller flow rate (back-face leakage flow rate plus flange flow rate) was scaled from the prototype and could be set to a desired value for a particular flow coefficient by adjusting the flow rate in the 2 leakage routing lines. Back-face leakage re-enters the flow path upstream of the test article inlet.

Five test series were conducted to fully document pump performance at each tip-clearance. These series included definition of the basic head-flow curve at constant suction specific speed, followed by complete suction performance mapping across a wide range of flow rates. Table 5 summarizes the test matrix and range of set point parameters.

Steady-state measurements acquired during testing were used to confirm test conditions, evaluate test article performance, and monitor test article health. Surface static pressure taps are distributed throughout the test article and are grouped into 27 measurement planes. Total pressure probes are located in the facility inlet and exit spools. Flow direction probes are located just downstream of the inlet guide vanes and impeller discharge for measurement of inlet and exit swirl. In addition to the steady-state measurements, unsteady pressures and accelerations were recorded during inlet pressure and speed ramps at each flow coefficient.

Test Results

Test article performance was calculated from measured values and plotted for comparison using a spreadsheet. Selected performance plots comparing the three tip-clearances tested are shown. Impeller head coefficient versus suction specific speed at design impeller flow/speed ratio appears in Figure 4, Impeller head coefficient versus normalized impeller flow/speed ratio appears in Figure 5, and Impeller head coefficient versus normalized impeller flow/speed ratio appears in Figure 6.
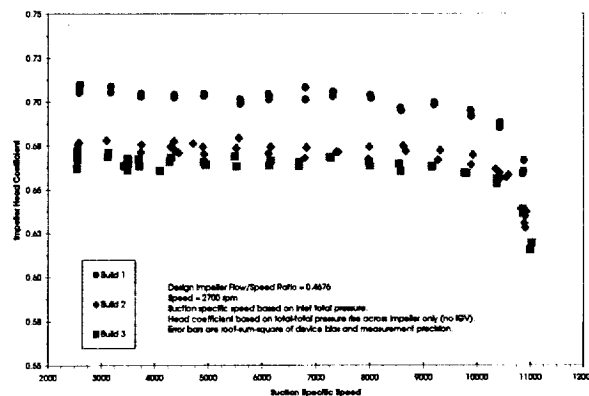

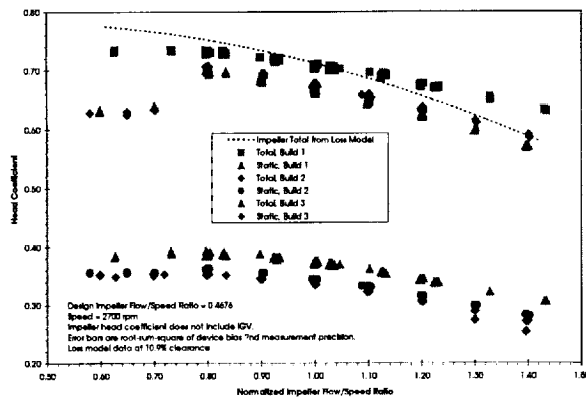Figure 4 - Impeller head coefficient versus suction specific speed at design impeller flow/speed ratio.

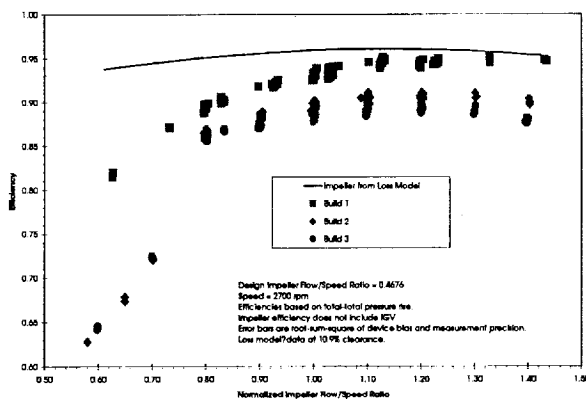**Figure 5 - Impeller head coefficient versus normalized impeller flow/speed ratio.**

Figure 6 - Impeller efficiency versus normalized impeller flow/speed ratio.

RLV Impeller Trade Study

The objective of this project was to develop a 2-stage unshrouded impeller that can be used to replace a 3-stage shrouded impeller in the RLV engine. The net result would be a lower weight RLV turbopump design which has significant impact on the engine thrust to weight requirements and would not compromise rocket engine performance (Isp). One concern with unshrouded impellers is the loss of performance due to tip-clearance. If the performance loss is great then the turbine must produce more horsepower and consequently an increase in either turbine temperature or turbine mass flow must be allotted. This obviously would have negative impacts to the turbopump and engine design. The goal of the design trades was to develop an understanding and achieve an impeller design that minimizes the performance sensitivity to tip-clearance.

A number of papers have been published on the impact of tip-clearance on unshrouded impellers for compressors and pumps. In 1972 Rocketdyne

completed an evaluation of shrouded and unshrouded impeller performance using the J2 liquid oxygen turbopump. Water tests were completed with a shrouded impeller and an impeller with the shroud removed at various tip-clearances. The performance impact of a 10% increase in the tip-clearance resulted in a 12% decrease in efficiency[3]. Y. Senoo[4] wrote in 1987 that a tip-clearance change from 0% (shrouded) to 10% of the impeller exit width decreases efficiency by 4% for compressors. The different impact of tip-clearance on these turbopumps indicates that impeller design parameters can impact the efficiency defect. In 1997 Johannes Lauer, et. al.[5] describes an experimental study of 14 semi-open (unshrouded) impellers of different designs. The results were not conclusive, but indicated that the blade number and exit angle had the largest impacts on tip-clearance sensitivity. Other items of interest in unshrouded impellers are the prediction of axial thrust as clearance varies, the existence of a hydraulic couple due to difference in front and rear pressure forces, and potential for rotordynamic forces which could destabilize the rotor.

Design Parameters

Decreased performance sensitivity to tip-clearance is a necessity to allow for incorporation of unshrouded impeller technology into rocket engine turbopumps. Based on literature review[6] and tip-clearance modeling assumptions, it was decided that the primary design parameters of interest are:

1. Blade solidity
2. Blade number
3. Blade wrap
4. Axial length
5. Diffusion factor
6. Cant angle
7. b_2-width
8. Exit blade angle
9. Head coefficient

Further review of these parameters indicated that three were fixed due to engine balance constraints or need to minimize changes to the tester. These are 1) Head coefficient, 2) Axial length (shroud contour), and 3) b_2-width.

With these parameters fixed, blade solidity, blade wrap, diffusion factor, and exit blade angle are all varied with change in blade number. This leaves blade number, and cant angle as the remaining parameters to study. Cant angle, having mostly second order affects on performance, was eliminated from the study. Cant angle could have a significant impact on structural

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design to meet increased tip speed and will be visited later. Blade number was the parameter selected for further study. Three impeller designs were completed as part of the blade number trade study to evaluate performance sensitivity: 5+5, 6+6, and an 8+8 impeller. Each impeller meets the performance goals of the RLV YRS-2200 Engine balance. Table 6 shows the design parameters for each of these designs.

Table 6 - Blade number design parameters.

Parameter	Blade Number		
	5+5	6+6	8+8
Head Coefficient	0.53	0.53	0.53
Exit Flow Coefficient	0.128	0.118	0.117
Diffusion Factor	0.80	0.60	0.43
Inlet Blade Angle, Deg.	22	22	22
Inlet Blade Height, Inch	1.6	1.6	1.6
Tip Diameter, Inch	8.0	8.0	8.0
b_2-Width, Inch	0.58	0.58	0.58
Exit Blade Angle, Deg.	74	49	38
Total Blade Wrap, Deg.	52	98	120
Axial Length, Inch	2.08	2.08	2.08
W_2/W_1 (Rel. Vel. Ratio)	0.88	0.90	0.90

Trade Study Results

Preliminary sizing codes were used in conjunction with three-dimensional (3-D) computational fluid dynamics (CFD) analysis to calculate the head rise performance of the various unshrouded impeller designs. Sizing codes are well calibrated for shrouded impeller designs but not for unshrouded designs. A well-established database for tip-clearance loss coefficients, as being defined in this study, is required. Three-dimensional CFD models have also been validated for shrouded impeller designs and have shown good agreement with test data. The modeling procedure and methodology for an unshrouded impeller is similar to a shrouded impeller. This section presents a summary of the 3-D CFD performed during the design of RLV unshrouded impeller. RLV HPFTP rated design point flow rate is 20,164 gpm at 32,000. Performance and tip-clearance sensitivity CFD analysis was performed over a range of flow rates between 80% and 120% design flow.

3-D Flow Models

The full 360-degree geometry was partitioned into a single periodic channel representing 1/5th, 1/6th, or 1/8th of the total impeller pitch, for the 5+5, 6+6, and 8+8 impellers, respectively. Impeller tip-clearances of 0% (shrouded), 6%, 10%, and 20% of the b_2 width were examined. The inlet (radial) and b_2 (axial) tip-clearances were varied separately.

Grid Generation

The numerical flow grids were generated algebraically from the impeller contour and surface definition. The grid generation tool was integrated with the impeller geometry tool to support quick parametric CFD analysis studies. Each unshrouded impeller model consisted of 7 zones. The grid dimensions are indicated in Table 7.

Table 7 - CFD grid dimensions.

Zone ID	Nodes (Meridional x Radial x Blade-to-Blade)
1	7 x 11 x 33
2	11 x 11 x 29
3	27 x 11 x 13
4	27 x 11 x 13
5	11 x 47 x 33
6	23 x 11 x 33
7	43 x 9 x 33

The shrouded impeller models (i.e., 0% gap) contain zones 1-6. Zone 7 was the tip-clearance region and the physical dimension from blade tip to shroud surface varied depending on the percent clearance analyzed. A systematic grid sensitivity study was not done since the objective was just to get performance trends. Figures 7-9 shows the flow grids for the 5+5, 6+6, and 8+8 unshrouded impellers with 10% tip-clearance.
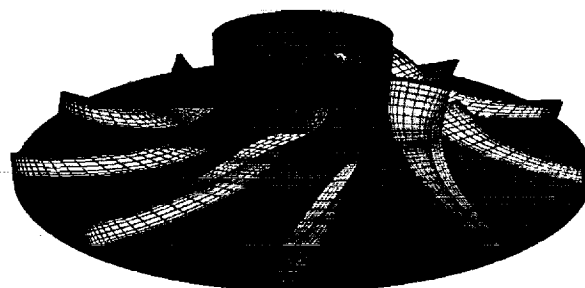

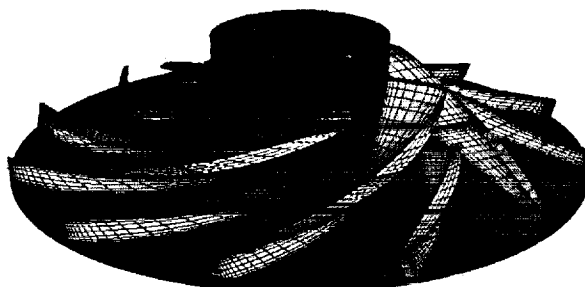**Figure 7 - 5+5 Impeller CFD surface grid.**

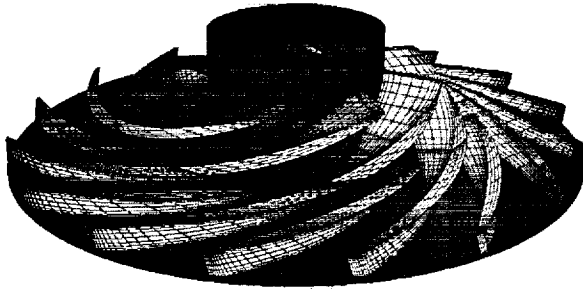**Figure 8 - 6+6 Impeller CFD surface grid.**

Figure 9 - 8+8 Impeller CFD surface grid.

The 5+5 geometry has the largest discharge blade angle, and 8+8 has the smallest discharge blade angle. A plot of blade angle distribution along a blade streamline is shown in Figure 10.
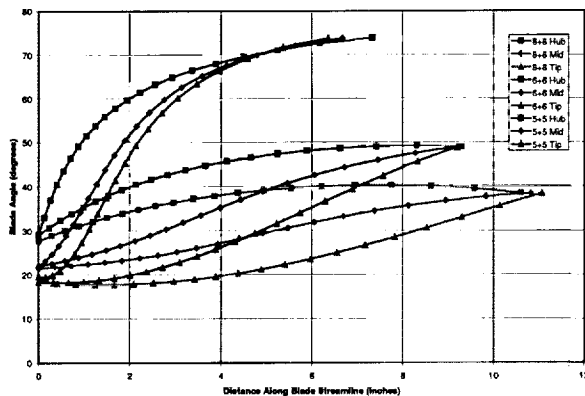

Figure 10 - Blade angle distribution.

CFD Parametric Studies

A parametric study of all three geometries was performed using CFD analysis. Over 60 CFD analyses were completed. Each geometry was analyzed at 0%, 6%, 10%, and 20% clearance. Each clearance was analyzed at on- and off-design conditions from 80% to 120% flow. Static pressures along the blade passage flow surface obtained from the CFD models were applied to a finite element model to determine blade stress. Pressure loading on the shroud surface was also used to determine axial load applied to the bearings. Figures 11-13 show static pressure contours for the 5+5, 6+6, and 8+8 unshrouded impellers with 10% tip-clearance.
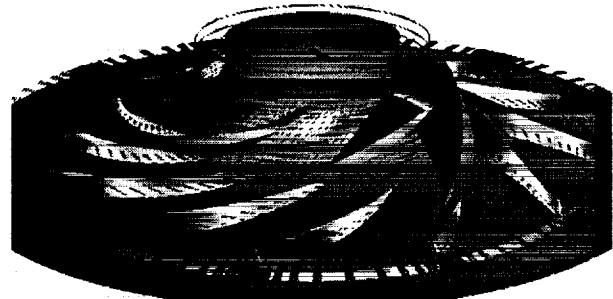
NRA8-21 High Head Unshrouded Impeller Pump Technology
Design2 5+5 Unshrouded Impeller C=0.063 H=0.062 - case100b
Relative Velocity Vectors & Blades Colored by Pressure

RLV H2 2-Stage Impeller - 32000 rpm, 209 lbm/sec, b2=0.58

Figure 11 - Blade pressure contours 5+5 Impeller.

NRA8-21 High Head Unshrouded Impeller Pump Technology
Design2 6+6 Unshrouded Impeller C=0.084 H=0.083 - case100b
Relative Velocity Vectors & Blades Colored by Pressure

RLV H2 2-Stage Impeller - 32000 rpm, 209 lbm/sec, b2=0.58

Figure 12 - Blade pressure contours 6+6 Impeller.

NRA8-21 High Head Unshrouded Impeller Pump Technology
Design2 8+8 Unshrouded Impeller C=0.063 H=0.062 - case100b
Relative Velocity Vectors & Blades Colored by Pressure

RLV H2 2-Stage Impeller - 32000 rpm, 209 lbm/sec, b2=0.58

Figure 13 - Blade pressure contours 8+8 Impeller.

A comprehensive database was compiled and results between cases were compared. Global performance parameters and local flow uniformity was assessed to determine the best candidate geometry for the 2-stage RLV HPFTP design. Figures 14 and 15 show impeller

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developed head and efficiency comparisons for each of the three geometries for three tip-clearances at the impeller design point.
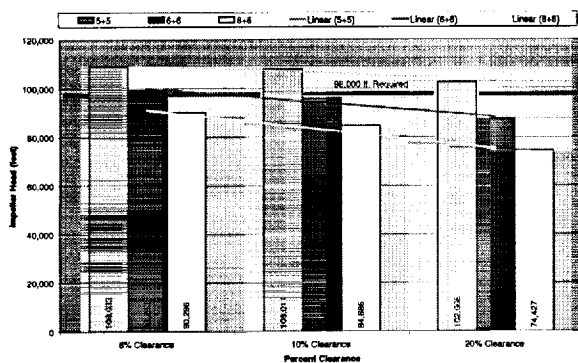

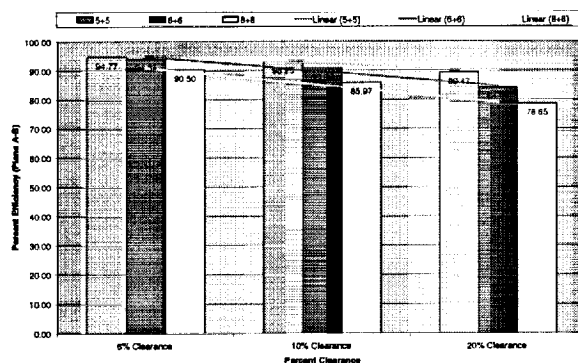Figure 14 - Impeller developed head.

Figure 15 - Impeller efficiency.

As expected, global performance parameters (head and efficiency) were greater for the 5+5 design due to more work performed on the fluid with greater blade turning and less viscous loss with shorter blade length. Additionally, performance decreased due to increased secondary flow loss with larger clearance. Locally, however, the 5+5 design's aggressive blade turning produced less desirable flow non-uniformity. Figure 16 shows the 5+5 design had substantially greater back flow at the blade leading edge.
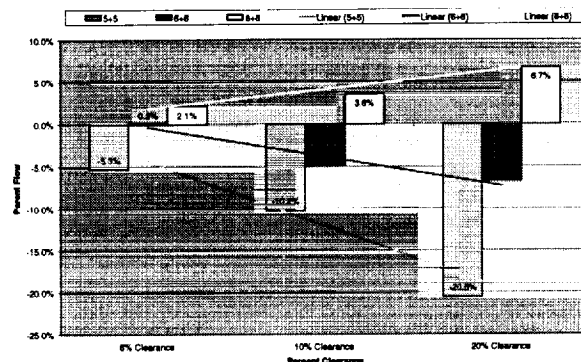

Figure 16 - Back flow at the blade leading edge.

The 6+6 design was selected for the RLV HPFTP 2-stage unshrouded impeller configuration based on overall performance and flow uniformity. The 6+6 design at 6% clearance also reached the 98,000-ft. developed head requirement per pump stage. The objective of this study was accomplished with the 6+6 impeller design. This technology will enable higher T/W engines and ultimately increase payload to orbit.

Vehicle system trades were performed to determine the overall potential increase in payload to orbit. The 6+6 unshrouded impeller design at 6% clearance had similar performance to a shrouded design. At this clearance the increase in payload per engine would be 860 lbs. (see Table 1 and Table 19). With larger operational clearances however the turbopump efficiency begins to decrease. Rocket engine specific impulse (Isp) also begins to decrease. This reduction in rocket engine performance decreases the potential increase in payload. Figures 17-19 illustrate the change in impeller efficiency, decrease in engine Isp, and increase in payload per engine for three geometries and three tip-clearances.
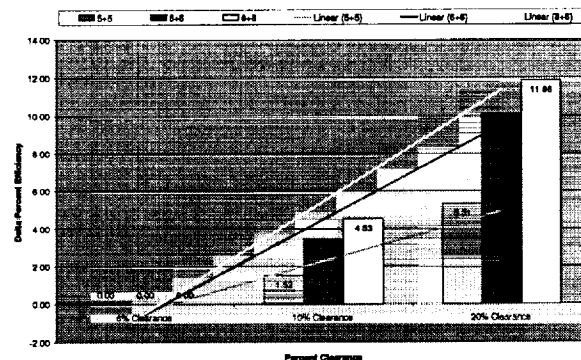

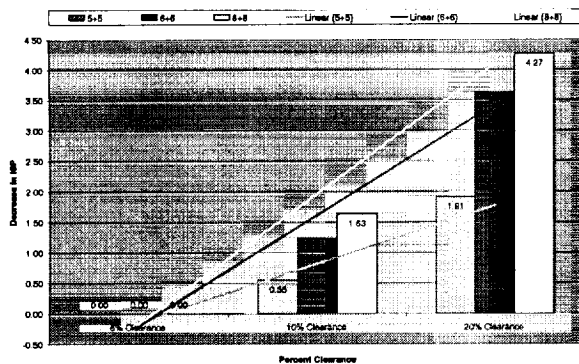Figure 17 - Change in impeller efficiency.
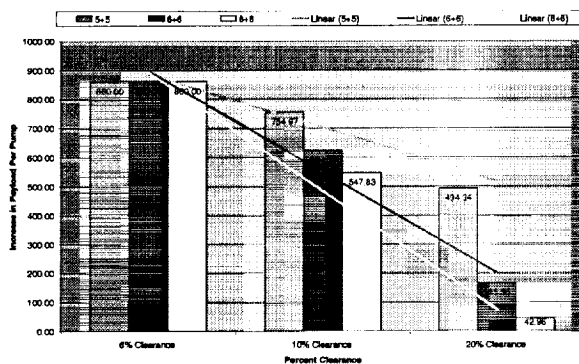
Figure 18 - Decrease in engine Isp.

Figure 19 - Increase in payload.

RLV HPFTP Mechanical Layout

A preliminary 2-stage turbopump mechanical layout was completed based on the RLV engine balance[7]. An inducer was sized to meet the requirements of the balance. Assuming long life goals, hydrostatic bearings were baselined. A clutching bearing was integrated into the layout to allow for transient start and shutdown loads. Wear rings and an inter-stage seal were defined to balance axial thrust and provide rotordynamic stability. The turbine envelope definition was based on an advanced turbine under development in a parallel NRA8-21 project. Axial length, based on design rules, provides spacing between turbine and pump to accommodate turbine temperatures defined in the governing balance.

An SSME style rotor stack up was used on the turbopump rotor to ensure adequate preload during assembly, chill, and operation and to maximize rotor stiffness for rotordynamics. The rotor stack up and balance piston was designed to allow for turbopump operational tip-clearance between 6% and 12% of the impeller discharge blade height. The preliminary design also included assessment of axial thrust, rotordynamics, weight, and impeller stress to ensure a viable concept to advance to an operational turbopump.

Conclusions

The objective to develop an unshrouded impeller design, which meets the performance requirements of a 3-stage fuel pump with a 2-stage pump design, has been accomplished. The performance of the baseline unshrouded impeller has been experimentally verified. The unshrouded impeller trade study and final 6+6 unshrouded impeller configuration has been presented. A structurally viable, 6+6-impeller design concept has been produced. Based on results presented in this study, at a nominal 10% tip-clearance, the 6+6 impeller design would increase payload to orbit by almost 625 lbs. per engine (see Table 19). The RLV vehicle requires 7 engines. Therefore, application of high head unshrouded technology would increase payload capability by as much as 4,375 lbs. per vehicle.

References

[1] Williams, R., "Project Plan for High Head Unshrouded Impeller Pump Stage Technology in NRA8-21," Memo ED32(98-027), July, 1998.

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[3] R.K. Hoshide and C.E. Nielson, "Study of Blade Clearance Effects on Centrifugal Pumps," Contract #: NAS3-13311, Report #: NASA CR-120815, NASA-Lewis Research Center.

[4] Y. Senoo and M. Ishida, "Deterioration of Compressor Performance Due to Tip Clearance of Centrifugal Impellers," Journal of Turbomachinery, January 1987, Vol. 109, pp. 55-61.

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[7] G. H. Prueger, et. al., "Milestone #4, Unshrouded High Head Impeller Preliminary Layout," Contract #: NAS8-98259, High Head Unshrouded Impeller Pump Stage Technology, May 2000.

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